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No. 1069

THE FRICTION OF PISTON RINGS

By Hans W. Tischbein

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THE FRICTION OF PISTON RINGS*

By Hans W. Tischbein

The coefficient of friction between piston ring and cylinder liner was measured in relation to gliding acceleration, pressure, temperature, quantity of oil and quality of oil. Comparing former lubriction-technical tests, conclusions were drawn as to the state of friction. The coefficients of friction as figured out according to the hydrodynamic theory were compared with those measured by tests. Special tests were made on "oiliness." The highest permissible pressure was measured and the ratio of pressure discussed.

SUMMARY

The coefficient of friction between piston ring and cylinder wall (both average coefficient and coefficient at different points over the stroke) was measured on a test setup in relation to rubbing speed, wall pressure and temperature, increase in running-in time, oil quantity, and type of oil. The average friction coefficients fluctuated between 0.02 and 0.14 and, for the same oil samples and test conditions at medium and high wall pressures, was much greater than the friction coefficients of a well-oiled journal bearing.

Mixed friction was, in general, found to exist, except at low wall pressures, over working-surface temperatures with high average rubbing speed, where fluid friction in the central part of the stroke may be considered likely.

The friction coefficients calculated for the present case in support of Gümbel amounted to a multiple of the experimental values. According to this the premises underlying these theoretical calculations appear to be wrong. The influence of oiliness was discernible. For the same test conditions, different oils of the same viscosity showed different coefficients of friction. The widest discrepancies occurred on

*"Reibung an Kolbenringen," Kraftstoff, Dec. 1939, pp. 83-87, Jan. 1940, pp. 6-8, Feb. 1940, pp. 39-42, and March 1940, pp. 71-75.

the engine oils; whereas the bearing oils disclosed only minor differences. A comparison of the test data on the basis of Vogelpohl's number indicated that this number affords no adequate explanation for the differences in friction coefficient.

The limits of the highest permissible wall pressure of the piston ring on the liner and the highest permissible working-surface temperatures also were measured, and the pressure conditions at the piston ring discussed, particularly the wall pressure due to gas pressure and moving-away of the ring as a result of the reciprocal action of frictional force and gas pressure.

INTRODUCTION

In the friction of two solid surfaces sliding over each other a differentiation usually is made between dry, fluid, and mixed friction.

In fluid friction the rubbing surfaces are completely separated by a fluid film. The lubricant adheres to the material of the rubbing surfaces, and the entire friction process takes place in the lubricating film between the two surfaces. The frictional resistance is, therefore, due to pure fluid friction. In laminar flow the shearing stress in the fluid is, according to Newton, equal to the product of velocity gradient and dynamic or absolute viscosity. Reynolds (reference 1) applied this theorem to fluid friction, while Gümbel (reference 2) extended it to include journal bearings. This theory has been largely confirmed by experiment, but there are discrepancies also. Thus, oils of different source and treatment exhibit, in spite of identical viscosity, different coefficients of friction in the journal bearing. Voigtländer (reference 3) and Buche (reference 4) controlled the viscosity in their tests on dissimilar oils by varying the oil temperature and found discrepancies up to 25 percent. An explanation for the discrepancies is principally looked for in the molecular-physical sphere. (A complete survey of modern views on surface condition and friction is given in a book by Schmaltz (reference 5) which also contains a very comprehensive list of references. Kyropoulos (reference 6) has listed the physico-chemical properties of lubricating films. A detailed catalogue also is given.) The solid surfaces sliding over each other are carriers of free valences. In this field of force the oil molecules stretched out at full length

with their polar ends are adsorbed to the surface and set up at right angles to it. Depending upon the field of force and the chemical composition of the lubricant molecule one or more of such adsorbing layers cover the surface of the solid body. The layers slide over each other, whereby the inactive ends function as sliding surfaces. During the motion, the molecules at right angles to the surface are obliquely bent. But aside from these adsorbing layers the arrangement of the molecules themselves in the lubricating layer between these boundary layers is of influence on the friction. The long lubricant molecules orient themselves with their longitudinal axes in flow direction (flow orientation) and so reduce the frictional resistance in the fluid layer, since the dynamic viscosity at flow orientation of molecules is lower than the values recorded with a viscosimeter. This viscosity reduction depends upon the chemical structure of the lubricant molecules, the rubbing speed, and the clearance width. These influences together with the conditions for the adsorbed boundary layer produce differences in the frictional forces and therefore may explain a discrepancy of the computed values from those obtained according to the theory of fluid friction.

A further reason for the discrepancy between the theoretical and experimental data is to be found in the fact that the theory of fluid friction premises perfectly smooth rubbing surfaces. But the surface roughness can be of the same order of magnitude as the clearance width (reference 5) and in that case is not negligible with respect to the oil-film thickness.

If the distance between the solid bodies is very small, the adsorbed boundary layers slide directly over each other. When the surfaces are flat and separated from each other by a film of lubricant of only a few molecules, the friction is that of a boundary lubrication. In contrast with fluid friction, boundary lubrication can be associated with wear, because the lubricant molecules adhere so strongly to the wall that parts are torn out of it. The coefficients of friction for boundary lubrication are substantially greater than for fluid friction. Pure boundary friction conforming to definition is practically unattainable because of the roughness of the rubbing surfaces.

According to older concepts the semi-fluid or mixed friction is the simultaneous appearance of fluid and dry friction by thin lubricating film. But the existence of absolutely clean surfaces lacks empirical basis according to more modern concepts; adsorbed lubricant layers of special structure are always present at the surfaces. At present, mixed friction

defines the zone in which the fully fluid friction changes to pure boundary friction. The mixed friction is predominately associated with wear, because the surface pressures between the prominent roughness peaks can presumably become so great that these particles can be sheared off or torn out without interrupting the adsorbed lubricant film at the point of the fracture. The greater the proportion of boundary friction to the mixed friction, the greater the effect of those factors which are not included in the theory of fluid friction. Exploration of the boundary film indicates that the friction cannot be adequately explained on the basis of purely mechanical considerations. The multitude of carefully carried through exploratory labors yield, as such, much valuable material for the explanation of the lubricating process, but no definite criterion ever has been found for the so-called oiliness. Numerical data on properties such as the heat of adsorption, orientation of flow, capillary constant, dielectric constant, and so forth, yield a sense of direction - for example, it has been determined that the heat of adsorption decreases with increasing friction (reference 4) - but the experimental conditions are not always definite for a comparison.

A new method for judging the oiliness is given by Vogelpohl (reference 7), who sees an explanation for the discrepancy of the friction tests in the viscosity distribution in the lubricating film. The viscosity in the direction of motion in a plain bearing decreases as a result of the temperature rise in the lubricating film. There also are differences in viscosity in radial direction. The decrease in viscosity with respect to a specified average value is a measure for the carrying capacity of the bearing. Vogelpohl forms a characteristic for the oiliness, which contains the temperature relationship of the viscosity, the specific weight, and the specific heat.

In a comparison with Voigtländer's bearing friction tests he finds that the characteristic increases with growing frictional force, but in view of these few tests his theory does not appear reliable enough and requires additional tests for confirmation.

The appraisal of oiliness of oils from different sources and with different pre-treatment still necessitates comparative tests, with the test conditions as closely as possible adapted to service conditions.

Past Experiments on Piston-Ring Friction

The majority of lubricating research is concerned with bearing friction; whereas piston-ring friction has been considered very little so far. Its research is rendered difficult by the fact that the surface pressure between the rubbing surfaces is usually not known, because, for example, it is affected by the working pressure in the cylinder.

T. E. Stanton (reference 8 pp. 469-472) first measured the friction between piston and piston rings on a special test setup. It was found that the piston-ring friction increased very little with increasing gas pressure over the piston. The gas pressure in the ring grooves was also measured.

Mader (reference 9) who also used a special setup for his piston-ring friction studies found, like Stanton, a slight increase in frictional force with the gas pressure. His tests showed that the frictional force increased proportionally to the number of rings.

Ricardo (reference 10) measured the frictional force of pistons and rings in relationship to cooling-water temperature and the number of rings on an electrically driven internal-combustion engine. He found the friction force to decrease with increasing temperature and the increase proportional to the number of rings.

Vogel (reference 11) in his leakage tests on a Diesel engine measured the gas pressure between the first and second piston ring. He also investigated the friction values of the rings, without, however, measuring the frictional force directly.

A number of other articles deal with the natural stress of the piston ring, with the heat flow through the piston ring, or else touch the problem of piston ring friction in conjunction with other investigations.

The reports mentioned so far afford a partial explanation of the effect of the gas pressure on piston-ring friction, but fail to give sufficient information from the point of view of lubrication. An article by Eweis (reference 12) published while the present investigation was under way - (various preliminary results have already appeared in the following report: Aus deutschen Forschungstatten (From German Research). Archiv für Wärmewirtschaft und Dampf-

kesselwesen, vol. 16, Heft 1, 1935, pp. 19-20) - contains a discussion of the friction between piston ring and cylinder wall, of dry and semi-fluid friction, and a calculation of the frictional force for sharp-edge and rounded-edge rings under fluid friction, along with a numerical prediction of the supposed pressure distribution of the gas at the back of the rings. The pressure distribution and the friction between ring and cylinder wall were checked by test. The pressure distribution was found to be in good agreement with the theory. The effect of the gas pressure above the piston and in the groove in back of the rings was determined, the relationship between frictional force and number of rings investigated for self-expanding, non-gas-loaded rings, and also for rings subjected to gas pressure constant and variable with respect to time. Eweis' principal finding was that the friction of piston rings without gas loading was proportional to the number of rings. Under constant gas pressure the friction is almost proportional to the gas loading, but under variable gas pressure over the bottom of the piston the increase in friction is greater with few than with many rings. However, the state of friction of the piston rings requires further explanation. It still is uncertain at what operating conditions the fluid or the semi-fluid friction prevails and whether it is permissible to figure with the hydrodynamic theory in the prediction of the frictional force. (Salzmann, for instance (reference 13), used the hydrodynamic theory in his study of the heat flow through the piston ring to determine the thickness of the lubricating film.)

The present report is chiefly concerned with purely lubricating problems. The extent to which the high gas pressure in the combustion chamber and in the piston-ring groove governs the pressure of the ring against the cylinder wall is not quite clear from the investigations made hitherto. Some supplementary considerations concerning this question are included.

EXPERIMENTAL SETUP

Because it did not seem feasible to measure the piston-ring friction on a running engine, a special test ring was used. It was very important that temperature, wall pressure, and rubbing speed be those obtained in normal service as much as possible. In order to explore the effect of these several influences on the piston-ring friction it was necessary to design the test setup so that the quantities such as rubbing

speed, pressure per unit of area, working surface temperature, and amount and kind of lubricant could be varied individually and independently of each other. It was also of importance that, aside from the average friction values, the instantaneous values of the friction coefficient variable with the piston stroke, could be measured.

The movement of the piston was produced by a machine crank linkage (part of an erstwhile horizontal gas engine) of 210 millimeters stroke. The diameter of the liner was chosen at 205 millimeters. The piston rod of the test piston was supported at both sides of the liner and assembled with radial clearance to the cross head of the gear. The crankshaft was driven by a belt from an electric motor.

The liner was fitted with a heating coil so that its temperature could be regulated between room temperature and 200° C. The temperature of the liner was measured with five thermocouples suitably spaced 0.5 millimeter below the liner.

The lubricant was supplied in the middle of the liner through five radial holes; the amount was regulated by a drop oiler.

Through the reciprocating motion of the piston, the piston rings are moved over the liner at a rubbing speed varying from zero to a maximum value, whereby each time the state of "acceleration" and "deceleration" must be passed through. If the feed of the lubricant proceeds through the liner or from the crankcasing, the state of lubrication varies according to the distance from this feeding point.

From one to four rings could be fitted in the piston; the side clearance mounted to 0.05 millimeter. The rings were 4.5 millimeters thick and of 10 millimeters width. They were not self-explanding but pressed against the sliding path by a special expanding device (as suggested by Walger). (See fig. 2.) Two double levers C with uneven arms pivoted on knife edges were pulled together at the longer end by a spring D and so exerted at its short-lever end an expanding force on the face of the ring end. This force T is computed from the spring force and the lever system. With F indicating the face area of the piston ring, the radial wall pressure p_a introduced by this expanding force is

$$p_a = \frac{2T\pi}{F} \ (kg/cm^2) \qquad (1)$$

on the assumption that the outside diameter of the ring in the unexpanded state was equal to the diameter of the liner. The tension of the spiral spring D as a measure of the expanding force T was read on a millimeter scale.

The first test ring is shown in figure 1. The cast-iron liner, open at both ends, was suspended from wires, permitting small motions in direction of the longitudinal axis practically without resistance. A special type of piston carried the piston ring through the liner so that only the ring touched the liner. Owing to the frictional force between ring and liner, this likewise goes through a reciprocating motion because of the reciprocating movement of the piston. The time-displacement curve of the liner was plotted at enlarged scale from a photographic recording device. The frictional force between ring and liner follows from the time-displacement curve of the liner motion according to

$$P_r = Mb \text{ (kg)} \qquad (2)$$

where M is the mass of the liner and b the acceleration imparted to the liner by the friction. The coefficient of friction which, by definition, is the ratio between frictional force and normal force then follows from

$$\mu = \frac{P_r}{p_a F} = \frac{Mb}{p_a F} \qquad (3)$$

The speed and the acceleration as well as the corresponding frictional force with respect to time were obtained by graphical differentiation from the displacement-time curve of the working surface; the frictional force then was plotted against the piston displacement by means of the known relationship between displacement and time. Preliminary tests with this setup, however, indicated that the graphical differentiation for predicting the acceleration and the friction value was too inaccurate, so that the setup was redesigned for a different measuring system. The previously easily movable liner was damped between four very stiff springs and the deflection of one spring recorded. This is a practical measure for the frictional force exerted radially on the liner, if the motions of the liner are so small that its mass forces can be discounted. After initial difficulties it succeeded in placing the natural vibration frequency of the elastically suspended liner very much higher than the frequency of the piston drive in spite of its great mass.

Prerequisite of the accuracy of the evaluation of the pressure-time record was a natural vibration frequency of at least 400 Hertz. This was obtained by extremely stiff springs imbedded in a solid concrete foundation. Since the actually recorded natural vibration frequency was consistently much lower, usually only half of that computed from the spring dimensions, the elastic restraint of the springs in concrete was looked upon as cause for it. The flexibility of the base was then measured by mechanical-optical means, the reduction in the natural vibration frequency computed according to Klotter (reference 14) and Hayashi (reference 15), and a very satisfactory agreement with the measured vibration frequency ascertained.

The final version is illustrated in figure 3. The knife edges F mounted at the front of the liner on a level with the center of the liner rested in sockets of the flat springs G secured to the base and so supported the liner. Equal initial tension of all four springs was insured by spacers between sockets and springs. The initial tension was always greater than the maximum frictional force because the liner was so mounted that the reciprocating force was not transmitted to it.

The stiff springs kept the axial motion of the liner very small, the maximum deflections being only a few thousandths of a millimeter, thus necessitating a special method to measure them. The deflection of spring G was measured optically by a device illustrated in figure 4. One of the four springs extended beyond the socket B and carried a knife edge C at the end. This pressed on a small steel band which in turn was supported against a fixed knife edge D spaced horizontally about 0.2 millimeter from C. The steel band was under initial tension with respect to the knife edges by appropriate torsion and carried a tiny mirror F fastened with cement. When the spring twisted in consequence of the frictional forces exerted on the liner, the mirror turned and a light ray reflected by it was deflected. The path of this light ray recorded by a specially designed photographic recording device then immediately showed the time variation of the frictional forces with consideration to the calibration.

The calibration was obtained by exerting an additional known axial force on the liner and recording the deflection of the light ray on the recording instrument. To prevent irregularities arising from the naturally very small friction in the sockets, the calibration was made during the recipro-

cating motion of the piston and the displacement of the deflection noted on the slowly running film. The coefficient of friction μ is then computed from the relationship between frictional force P_γ and the expanding force T at the piston ring:

$$\mu = \frac{P\gamma}{2\pi T} \qquad (4)$$

Since the system susceptible to vibration was strongly excited by the oscillating motion of the piston, it became necessary to dampen the vibrations of the liner. For this purpose a larger mass i was freely suspended alongside the liner (fig. 3) and coupled to it; the contact surfaces, k and l, of the liner and damping mass respectively, were separated by an oil film.

To ascertain the extent of the existing fluid friction, 0.2 volt was applied to the piston ring and the liner, which, at metallic contact between ring and liner, or penetration of the oil film, caused a glow tube of special design to light up by means of interconnected amplifier tubes. The lighting and extinguishing of the glow tube was recorded on the strip along with the frictional force curve. However, the glow tube always continued to burn during the tests even though the other test results argued in favor of fluid friction. There must have been locally narrowly defined metal contacts, perhaps at the edges of the spreading area, even while the rest of the ring was in the state of friction. This arrangement was therefore unsuitable for judging the state of lubrication. It can only be stated that, at some point of the ring, the distance from the wall of the liner was always only of the order of magnitude of a few lubricant molecules.

With the new test arrangement, average speeds of from 0.25 to 3.5 millimeters per second, unit wall pressures from 0 to 13 kilograms per square centimeter and mean working surface temperatures from 40° to 250° C could be attained. The liner, with which the principal tests were carried through, was well run in during the preliminary tests of over 400 hours.

SELECTION OF PISTON RINGS AND OIL – TEST PROGRAM

In order to include the proportion of the friction that is not explainable by the viscosity in the tests, the

oils from different sources were always so selected that several oils had the same dynamic viscosity or nearly so at identical test temperatures. To this end a large number of commercial brands of oil were tested for viscosity and temperature relationships with the Ubbelohde viscosimeter (reference 16). The kinematic viscosity was measured up to 120° C, at higher temperatures the values were taken from the double logarithmic plot by Ubbelohde (fig. 5). The specific gravity necessary for calculating the absolute viscosity was measured at two temperatures. The assumption of linear relationship was sufficient for the present purpose.

Nine different oils were used in these tests. The commercial trade names and the manufacturers of the oils tested are given in table 1 with the absolute viscosity and the specific gravity of the oils at 50° C.

TABLE 1

ABSOLUTE VISCOSITY AND SPECIFIC GRAVITY OF OIL SAMPLES

Oil	Source	Absolute viscosity at 50° C (kg/sq m)	Specific gravity at 50° C (gr/cm^3)
1	Shell Voltol I; Rhenania-Ossag Mineral Oil Works A-G	0.0023	0.879
2	Shell BA 78; Rhenania-Ossag Mineral Oil Works A-G	.0023	.880
3	Supplied by the Physical Technical State Institute for comparative tests (no other information available)	.0023	.8795
4	Supplied by the Physical Technical State Institute for comparative tests (no other information available)	.0023	.865
5	Shell Voltol II; Rhenania-Ossag Mineral Oil Works A-G	.0031	.879
6	Shell Rx; Rhenania-Ossag Mineral Oil Works A-G	.0086	.888
7	Valvoline, heavy; Valvoline Oil Co. Hamburg	.00745	.880
8	Mobiloil AF; German Vacuum Oil A-G Hamburg	.00695	.879
9	Shell 4x; Rhenania-Ossag Mineral Oil Works A-G	.0142	.8905

Oils 1 to 4 were light machine oils, 5 was a medium machine oil, 6 and 8 medium engine oils, 9 a heavy machine oil. The engine oils had been obtained from a filling station, the others from the manufacturers. Figure 5 gives the temperature relationship of the viscosity for these oils according to the formula by C. Walther. For clearness the kinematic viscosity was plotted in this representation. The tests always were adjusted for equal absolute viscosity. Oils 1 to 4 had the same absolute viscosity of 0.0023 kilogram per square meter at 50° C, oils 6 and 7 at 77.5° C, 6 and 8 at 98.8° C, and 7 and 9 at 149.7° C.

The working surfaces as well as the edges of the employed piston rings were machined in various manners. The surface was smooth-ground, fine-turned, or rough-ground; the edges were sharp, slightly or considerably rounded off. In the tests of oils 5 to 9, the wall pressure, working surface temperature, and mean rubbing speed were varied. Oils 1 to 4 were tested at only one wall pressure and one working surface temperature. The exact test conditions are given in table 2. The oil feed in all the tests was adjusted to 150 cubic centimeters per hour. The assembly contained only one ring, always of the sharp-edge smooth-ground type.

For oil 6, the running-in time was also determined. Furthermore, the relationships between coefficient of friction and ring-shape, number of rings, and quantity of oil were determined.

INTERPRETATION

Figure 6 is a force-time plot taken with the photographic recording instrument. The bright band is the path of the deflected spot of light. The time scale is given by a simultaneously recorded wave of 50 Hertz. The directional change of the piston is characterized by the pressure jump at the points "right dead center" and "left dead center." The attendant natural vibrations of the liner excited by the reversal of force at the dead centers are quickly damped out. By plotting the average position of the vibrations it was possible to approximate the variation of the frictional force in the vicinity of the dead center.

The frictional force was different for the forward and return strokes; no symmetry appeared to be present. The reason for this was that the ring received fresh oil in the

middle of the cylinder and hence exerted less frictional force in its path from the middle of the liner toward the dead centers than on its return.

For the determination of an average frictional force (average value during one reciprocal cycle) an arbitrary zero position was plotted (fig. 6), and the areas f_1 and f_2 on either side planimetered. Then the average of the friction coefficient was found from

$$\mu_m = \frac{f_1 + f_2}{h_1 + h_2} k \frac{1}{p_a F} \qquad (5)$$

where

h length of a stroke in the force-time diagram

k calibration factor, kilogram of frictional force per unit of deflection

p_a wall pressure

F rubbing area of the piston ring

Because the reversal in force at the dead centers is sudden, the sum of f_1 and f_2 is independent of the position of the arbitrary zero line.

For the determination of the friction coefficients variable with the piston displacement, the position of the light spot for the frictional force 0 in the force-time diagram was recorded directly. After taking a pressure curve with the recording instrument, the crank drive was immediately stopped, the piston ring relieved so as to prevent further frictional forces on the cylinder, and the position of the light spot photographed. The thus obtained zero lines appeared scattered on several photographs under identical test conditions, which could, for example, be attributed to temperature variations on the liner and on the test instrument in the time between taking the force-time diagram and the zero line. Since the direct measurement of the zero position was in consequence not certain, it was simply assumed that the frictional energy for the reciprocating motion and also for the areas f_1 and f_2 were of the same magnitude. The obtained zero line then corresponded on the average to the median position for the photographically recorded (scattered) zero lines.

REPRESENTATION OF TEST DATA

In the representation of the test data, the average value of the friction coefficient or of the frictional force over the piston stroke (hereinafter called mean friction coefficient or mean frictional force, for short) was plotted as ordinate and, in most cases, the average rubbing speed of the rings as abscissa. The friction coefficient or the frictional force for a certain piston setting was plotted against the respective rubbing speed, with the working surface temperature at constant wall pressure or the wall pressure at constant working surface temperature as parameter. But in several instances it seemed more advantageous to plot the friction coefficient against the temperature or viscosity, respectively, with the rubbing speed as parameter at constant wall pressure. Here the values for one diagram had been already obtained from another, a second graphical error compensation was omitted, and the points connected by straight lines.

EXPERIMENTAL RESULTS

Running-in time, piston-ring shape, number of piston rings, and quantity of lubricant.- The piston ring had to be run in in the liner before the tests started; it was considered run in when the friction coefficient showed no decrease after a protracted interval. During the running in of a piston ring the friction was measured from time to time. Figure 7 shows the decrease in friction with the time. After a short period the friction progressively decreased to a minimum which was about half of the initial value.

When a new ring was inserted, the outflowing oil turned dark for the first hours of the running-in process as a result of wear. A 1000-diameter magnification failed to disclose any particles under the microscope; hence the largest size of the particle should be less than 10^{-4} millimeter. The discoloration decreased with the running-in time; after the ring was run in there was no difference in color before and after use of the oil.

Piston rings with sharp and rounded-off edges (rounding-off radius 0.2 and 0.5 mm) disclosed almost the same decrease in friction coefficient with increasing running-in time, as shown in figure 7. Piston rings with sharp edges showed

higher friction coefficients at first than those with broken edges, but after a 20-hour running-in period under identical test conditions no appreciable edge effect remained. The running-in time of a smooth piston ring averaged about 40 hours. At a very low working surface temperature of 60° C and at a wall pressure not exceeding 0.1 kilogram per square centimeter the rounded-off rings yielded a smaller average friction coefficient than the sharp-edge rings. Furthermore, when the working surface temperature reached 215° C, the behavior differed according to the edge radius. Considerable lubricant evaporated from the surface at this temperature, leaving a viscous brown film that could not be scraped off from the ring with rounded-off edge and which increased the friction coefficient by 10 to 15 percent. The sharp-edge ring, on the other hand, consistently preserved a bright surface even at the highest temperatures.

The study further included piston rings with fine and coarse turning grooves on the working surface. The running-in time and also the initial coefficients of friction were greater than for the ground ring, and indeed increased as the coarseness of the turning marks increased. But after a suitable running-in time the friction coefficients were exactly the same for both.

Tests were made also with two and three sharp-edge, ground rings mounted simultaneously, at an average rubbing speed of 2.15 meters per second, a wall pressure of 3.0 kilograms per square centimeter, a surface temperature of 120° C, and an oil feed of 150 cubic centimeters per hour. At equal wall pressure, the frictional force was proportional to the number of piston rings.

To determine the effect of the lubricant quantity, special tests were made with oil feeds of 25 to 150 cubic centimeters per hour at average speeds of 1.5 to 2.80 meters per second, wall pressure of 3.0 kilograms per square centimeter, and surface temperature at 120° and 185° C. It was found that with very copious lubrication scarcely any relationship existed between friction coefficient and amount of feed; whereas starting at 40 cubic centimeters per hour the friction coefficient rose rapidly with decreasing oil quantity. At around 25 cubic centimeters per hour the quantity became insufficient and the surface started to change its appearance.

Experiments with oil sample 6 and comparison with hydrodynamic theory.- As seen from table 2, sample 6 was

studied at all test conditions. As the other oil samples could not be studied with the same completeness, the relationship between the coefficient of friction and the rubbing speed, the wall pressure and the surface temperature for the experiments with oil specimen 6 is discussed first and in connection herewith compared with the hydrodynamic theory.

Figures 8 and 9 show the average friction coefficient plotted against the average speed with the working surface temperature as parameter for two different wall pressures. In figures 10 and 11 the average friction coefficient is shown plotted against the average rubbing speed, with the wall pressure for two surface temperatures as parameter. The average rubbing speed was adjusted between 0.35 and 3.50 meters per second, the surface temperature between 65.3° and 215° C and the wall pressure between 1.5 and 9 kilograms per square centimeter. The average friction coefficients for all test conditions ranged between 0.02 and 0.13. In figure 12 the frictional force for wall pressures varying from 1.5 to 9 kilograms per square centimeter is directly plotted against the respective piston position. In this instance the average rubbing speed was 2.15 meters per second; the working surface temperature 119.6° C. The variation of the rubbing speed during the piston movement also is indicated. The frictional force is substituted here for the friction coefficient for the purpose of clearness. A noteworthy feature is the rapid rise in frictional force at low rubbing speeds in the vicinity of the dead center. This rise is so much more pronounced and the frictional force so much greater as the wall pressure and the working surface temperatures are higher.

On comparing, as in figure 13, the curve of the frictional force at different temperatures from 65.3° to 215° C and at the low wall pressure of 3.0 kilograms per square centimeter, it is rather unusual to see that at low temperature the maximum frictional force occurs in the center of the stroke rather than near dead center.

Figure 14 shows the frictional force plotted against the piston displacement at constant temperature of 63.5° C and constant wall pressure of 3.0 kilograms per square centimeter, but for average rubbing speeds of 1.15, 2.15, and 3.20 meters per second. Exactly as in figure 13, here also the maximum frictional force occurs at the center of the piston stroke.

The rubbing speed is a function of the piston setting and at the same time proportional to the rotative speed or mean piston speed. If, at any rotative speed, the rubbing speed is determined at each piston position and these rubbing speeds plotted against the coefficient of friction, the curves shown in figures 15 and 16 are obtained. The test points of each curve are shown for three mean rubbing speeds. Since the two halves of the stroke disclosed different friction coefficients for the same speed before and behind the oil hole as a result of the fresh oil reaching the ring in the center of the liner, only the friction coefficients for the piston motion from dead center to center of piston stroke were used for the representation. The curves show the relationship between frictional force and and rubbing speed, temperature and wall pressure; one unusual fact is that the frictional force despite the different average piston speeds is identically great at the same absolute piston speed. Fundamentally the variations of the curves in figures 15 and 16 are in agreement with those of figures 8 to 11.

The curves in figures 8 to 11, 15, and 16 manifest a certain similarity with the results repeatedly obtained for journal-bearing friction (references 17 and 18). For these tests (see fig. 19) the friction coefficient in the sphere of hydrodynamic friction customarily increases with the rotative speed; while the friction coefficient in the sphere of mixed friction passes through a minimum and then increases again considerably by decreasing rotative speed. Assuming that the relationship between speed and friction coefficient on the piston ring is, as for the journal bearing, a characteristic for the state of friction, some likely conclusions as to the type of friction might be drawn from these diagrams. It may be assumed that at the lowest running surface temperature at 65.3° C up to the wall pressures of 7.5 kilograms per square centimeter (fig. 10) hydrodynamic friction prevailed as yet, but that in all tests at substantially higher temperatures and high wall pressure (fig. 11) mixed friction predominates, as reflected in the more or less marked ascent of the friction coefficient with decreasing rubbing speed.

From figures 15 and 16 it may be concluded that mixed friction prevails near dead center even at low temperature and low wall pressure — that is, in the range of low rubbing speed — because the friction coefficients no longer decrease with decreasing rubbing speed toward the dead centers. Nevertheless, the curve of the "average" rubbing speed under identical test conditions is on the whole indicative of hydrodynamic friction. Thus the mixed friction seems to be relatively

small in the dead centers. But the higher the running surface temperature and the wall pressure become, the smaller the proportion of the fully fluid friction, even the range of maximum rubbing speed in the center of the piston stroke.

The friction coefficients of the piston ring can be computed by means of the hydrodynamic theory under simplifying assumptions. According to Gümbel (reference 2) a supporting oil film can be developed even between parallel rubbing surfaces if the front edge of the rubbing surface is rounded off. The rear edge plays a secondary part as proved by Salzmann. Assuming, according to Salzmann, flat rubbing surfaces and parabolically rounded-off edges, the equation for the average wall pressure p_a between the flat areas reads

$$p_a = \frac{3\pi\eta v}{8 h_o}\sqrt{\frac{\rho}{2 h_o}} \tag{6}$$

where

η absolute viscosity

v rubbing speed

h_o thickness of oil film

ρ radius of curvature in the apex of the parabola

Transformation of equation (6) gives for thickness of the oil film

$$h_o = \sqrt[3]{\left(\frac{3\pi\eta v}{8 p_a}\right)^2 \frac{\rho}{2}} \tag{7}$$

For the frictional resistance P_r of the sliding ring Eweis (reference 12) obtains

$$P_r = F \sqrt[3]{\left(\frac{8\sqrt{2}\, p_a}{3\pi}\right)^2 \frac{\eta v}{\rho}} \tag{8}$$

where F is the rubbing area of piston ring.

A further assumption is that the pressure before and

in back of the ring is zero in accord with the conditions of the previously described test arrangement.

After equations (7) and (8), in equation (3) the formula for the friction coefficient μ reads:

$$\mu = 1.13 \sqrt[3]{\frac{\eta v}{\rho p_a}} \qquad (9)$$

From equation (9) it is seen that in fluid friction the friction coefficient increases with increasing rubbing speed and decreases with rising wall pressure. Thus the existence of fluid friction is especially strongly indicated in the test data of figure 10, where the average friction coefficients increase rapidly with growing average rubbing speed and decrease with rising wall pressure.

Calculating, according to equation (9), the friction coefficient for oil specimen 6 at a rubbing speed of 3 meters per second, a surface temperature of 65.3° C, and a wall pressure of 1.5 kilograms per square centimeter while assuming ρ, say at 0.1 and 0.01 millimeter, respectively, gives the values of 0.218 to 0.470, as against the measured value of only 0.124. The calculated friction coefficient is therefore greater than the experimental coefficient.

In figure 15 the friction coefficient for 0.1 and 0.01 millimeter edge radii, 65.3° C surface temperature, and 3.0 kilograms per square centimeter wall pressure are plotted against the rubbing speed as computed by equation (9). Admittedly there is a considerable difference in magnitude, but at the low temperature of 65.3° C the variation of the curves for the theoretical and experimental friction coefficients is the same; hence, it may be deduced that in these test conditions the friction is predominantly fluid. One reason for the great difference is to be found, first of all, in the assumption of perfectly smooth rubbing surfaces. For the friction coefficients of 0.218 and 0.470 computed by equation (7) the oil film thicknesses can be predicted. They amount to 0.0033 and 0.0015 millimeter between ring and liner and hence have about the same magnitude as the roughness which runs to about 0.002 millimeter even for finely machined surfaces. Owing to the unavoidable roughness of the rubbing surfaces the oil film is thicker on the average than those computed previously, which might perhaps explain the inferior magnitude of the measured frictional forces. The marked effect

of the curvature radius in figure 15 could not be substantiated by tests. Piston rings with sharp edges and curvature radii of 0.1 and 0.5 millimeter were studied under otherwise identical test conditions and no measurable differences in friction coefficients were ascertained.

With a view to including as many test data as possible for a comprehensive comparison, the friction coefficient finally was plotted against the over-all characteristic

$$\varphi = \frac{\eta v}{p_a}$$

The test values were selected from the range where fluid friction was to be expected; that is, at low temperature and small wall pressure. Figure 17 shows that, throughout the entire range of characteristic theoretically computed friction, values are greater than the experimental values.

Experiments with different lubricants.- The exploratory tests with oil 6 indicated among other things that full fluid friction did not prevail during the whole stroke and that mixed friction always occurred in the vicinity of dead center. The proportion of the fluid friction in the mixed friction was as much smaller as wall pressure and temperature were higher. In this region of the mixed friction it is to be expected that viscosity and those other typical characteristics of oils called oiliness are especially evident. To preclude the viscosity effect in the subsequent tests the comparison was made with oils manifesting the same absolute viscosity at equal test temperature. Thus the light machine oils 1 to 4 had the same viscosity of 0.0023 kilogram per square meter at 50° C, and after minute comparative tests for these oils with the bearing testing machine in accordance with Walger-Schneider (reference 18) and Büche they then also were used for the tests on piston rings. The average friction coefficients were determined for rubbing speeds of from 0.2 to 3.0 meters per second at 1.5 kilograms per square centimeter constant wall pressure, 50° C temperature, and 150 cubic centimeters per hour oil feed. The difference between the individual oils, while being slight according to figure 18, nevertheless shows the sequence in the magnitude of the friction values very plainly. Oil 1 has the lowest friction coefficients, followed by oils 2, 4, and 3. The curves manifest a surprisingly great similarity to the curves

obtained with the same oils on the plain journal bearing. (Cf. fig. 19, where some of Büche's test data have been reproduced.) The oils indicated by A and B are the 5 and 2 of the present report. As on the plain journal bearing the friction coefficients on the piston ring decrease with decreasing rubbing speed, pass through a minimum, and then rapidly rise again with further decreasing rubbing speed. In the tests with the bearing-testing machine, oil 1 likewise had the lowest friction coefficient; then follow 4, 3, and 2; only 2 therefore shows a divergence.

Noteworthy is the difference between the lowest friction coefficients of a plain journal bearing of around 0.001 to 0.002 and those of a piston ring of about 0.05 to 0.10. The coefficient of friction of a piston ring is about 50 times greater than that of a journal bearing.

Aside from the light machine oils 1 to 4 the friction coefficients of the commercial engine oils 7, 8, and 9 also were determined with engine oil 6 included for the comparison. Oils 6 and 7 had at 77.5° C the same absolute viscosity of 0.00247 kilogram per square centimeter, oils 6 and 8 a viscosity of 0.00117 kilogram per square centimeter at 98° C, and oils 7 and 9 a viscosity of 0.000380 kilogram per square centimeter at 149.7° C. The friction coefficient was determined at 3.0 kilograms per square centimeter wall pressure and at the test temperature at which the same viscosities appeared. (See fig. 20.) It was found that in spite of equal viscosity at the same test temperature these oils also manifested a difference in friction coefficient. Thus, at a mean rubbing speed of 3.0 meters per second, the difference in the friction coefficients of oils 6 and 7 mounted to about 10 percent, and for oils 7 and 9 it was even greater at 1.5 meters per second mean rubbing speed.

It is surprising that the difference in friction coefficients of oils 6 and 7 is so great, although on the average, fluid friction might be presumed in view of the low working surface temperature of 77.5° C and the ascent of the friction coefficient with increasing speed. The friction coefficients of oils 7 and 9 show differences as high as 15 percent at 149.7° C and are indicative by reason of the speed relationship of mixed friction. On the other hand, the measurements on oils 6 and 8 at 98.8° C disclosed no appreciable difference in friction coefficient, although the aspect of the curves themselves is indicative of mixed friction.

For the comparison of oils 5, 6, 7, 8, and 9, throughout the entire viscosity range the average friction coefficients were plotted (fig. 21) against the absolute viscosity. The wall pressure was 3.0 kilograms per square centimeter and the average rubbing speed 3.0 meters per second. Although few in number at low viscosity, the test points for oil 9 fit quite well in the total picture. Oils 6, 7, and 8, which at the same temperature have approximately the same viscosity, manifest only minor differences in friction coefficient. Oil 5, on the other hand, has substantially lower temperatures throughout the whole viscosity range and exhibits a varying behavior. (See fig. 21.) In the zone of high viscosities, oil 5 shows lower friction coefficients than oils 6, 7, and 8; by decreasing viscosity it has, starting from 0.0025 kilogram per square meter, about the same viscosity, and from 0.0005 kilogram per square meter on, the friction coefficient increases rapidly with further decreasing viscosity.

Figure 22 shows the same representation for an average rubbing speed of 0.65 meter per second after an initial decrease with decreasing viscosity, the friction coefficients increase and reach a maximum which for oil 5 lies at η = 0.0012, for oil 8 at η = 0.00035, for oil 6 at η = 0.00032, and for oil 7 at η = 0.00028, after which they decrease again at very low viscosity. At low viscosity, oil 5 exhibits a rapid rise, which might be due to intense wearing; the outflowing oil turned dark in this viscosity range in the tests with oil 5. It may be presumed that a similar rise would occur in oils 6, 7, 8, and 9 also at lowest viscosity, but with the present test arrangement such low viscosities were unobtainable; hence the tests could not be extended to include these oils in the region of intense wear.

The cited maximum becomes much plainer as the rubbing-speed – that is, as the proportion of the boundary friction within the mixed friction – is greater. The appearance of the maximum seems to have a certain similarity with the experimental results by Herschel (report presented at the International Congress for Lubricating Research, Strassburg, July 20–26, 1931, a summary of which was published in VDI Vol. 75, pp. 1539–40). Herschel found on an oil-testing machine built by himself that at decreasing rubbing speed the friction coefficient varied similarly to figure 18. Having reached a minimum value, the friction coefficient rises rapidly, but then decreases again at considerably reduced rubbing speed. In the range of predominant mixed friction he also found that, with growing proportion of boundary friction to the mixed friction, the friction coefficient exhibits a maximum value in its curve. The result

of figures 21 and 22 can be summarized to the effect that the oils exhibiting only minor differences in viscosity at the same temperature, will show only slightly different friction coefficients throughout the whole viscosity range; while oils with widely varying viscosities at the same temperature manifest greater differences in friction coefficients and in their relation with the viscosity. The differences are much more pronounced as the temperature is higher and the rubbing speed lower, or in other words, as the proportion of the boundary friction within the mixed friction is greater. The tests indicate that the lubricating quality of an oil cannot be explained by the viscosity alone even in the range of predominantly fluid friction, in spite of the fact that the viscosity is one of the chief factors defining the friction.

Mention has been made in the foregoing of the attempts to explain this varying behavior of oil by means of the physical-chemical properties, such as adhesion of oil to the metallic surface, orientability of molecules in oil film, and molecular structure, for example. Because these properties can be influenced by additives, tests had been carried on in which oleic acid had been added to the lubricant (reference 19). (Also in unpublished tests of the machine laboratory of the Karlsruhe Technical High School.) Such an addition had no measurable effect on the piston ring. But an addition of colloidal graphite yielded a certain although slight reduction in friction, amounting to 6 percent in the zone of pure mixed friction, but no measurable decrease in the zone of presumable fluid friction. This finding is similar to that made by O. Walger and G. Schneider (reference 18) and O. Walger and H. V. Schroeter (reference 20) on journal bearings. Vogelpohl (reference 7) then attempted to explain these differences by means of hydrodynamics. (The experiments of the author had already been completed when Vogelpohl's article appeared in "Öl und Kohle." Subsequently his work was published in VDI-Forschungsheft No. 386.)

He proceeded from the fact that heat is created by the friction in the oil film, which frequently produces a substantial temperature rise. In accord with it he regards the decrease in viscosity resulting from the temperature rise as decisive for the carrying capacity of the oil film. Under equal test conditions, such as wall pressure, rubbing speed, and initial viscosity at entry of the oil in the oil film, the decrease in viscosity is much greater as the viscosity-temperature curve is steeper and the specific gravity and the specific heat of the oil are lower. He derives, for different

oils at equal wall pressure, equal speed, and equal initial viscosity, a relationship between the frictional force and a characteristic K, which is computed from the equation

$$K = \frac{\beta}{\gamma c} \tag{10}$$

where

γ specific gravity

c specific heat of oil

β a measure for the steepness of the viscosity–temperature curve

This can be represented in a certain viscosity range by the exponential function

$$\eta = \eta_1 e^{\beta(t - t_1)} \tag{11}$$

with η and η_1 denoting the viscosity values for temperature t and t_1, respectively.

The specific gravity and the viscosity of the oils were measured, the specific heat of the oils computed according to Kraussold (reference 21). (The values for γ, η, and c are in unusually good agreement with the data supplied by the oil companies for comparative purposes.)

As the comparison with Vogelpohl had to be made in the zone of the mixed friction, the characteristic K next was determined for oil samples 1 to 6, and 9 at a viscosity of 0.0023 kilogram per square meter, at 0.6 meter per second average rubbing speed, and 1.5 kilograms per square centimeter wall pressure. Depending upon the brand of oil, the viscosity corresponded to a test temperature ranging from 50° C to 186.5° C; β was determined, as by Vogelpohl, at the range between the initial viscosity and one-third of the initial viscosity. Figure 24 gives the calculated values of K and the related friction coefficients. While the values for oils 1 to 4 can be joined by a straight line, those for oils 5, 6, and 9 are widely scattered.

A comparison of oils 5 to 9 at a lower viscosity of

0.000380 kilogram per square meter gives a similar picture. (See fig. 24.) The test temperatures ranged between 124° C and 149.7° C, the wall pressure was 3.0 kilograms per square centimeter. At 0.6 as well as at 2.6 meters per second average rubbing speed, the engine oils 6 to 9 can be joined by a straight line; while oil 5 exhibits from time to time a considerable discrepancy. The curves in both diagrams indicate that the higher characteristic belongs to the greater frictional force. Vogelpohl found the same result for Büche and Voigtländer's test values.

In figure 23 the oil samples 1 to 4 had at 0.00230 kilogram per square meter the same test temperature of 50° C, the test temperatures for oils 5, 6, and 9 were 58° C, 79.8° C, and 86.5° C, respectively. In figure 24, oil 5 had a test temperature of 124° C, while that of oils 6 to 9 ranged between 140° C and 149.7° C. Thus it is readily apparent that guide lines can be plotted for a distinct relationship of the friction coefficient only when the oils have about the same viscosity at the same test temperature. If there are marked discrepancies, as in figure 21, Vogelpohl's data appear no longer sufficient to explain the differences. Furthermore, the effect of adsorbed layers also would have to be noted.

Limits for wall pressure and working-surface temperature.- Wall pressure and working-surface temperature may not be raised at will in the tests. At a specific wall pressure a certain working surface temperature could not be exceeded without entailing variations in the appearance of the liner. Because the liner was open at both ends, the rubbing surface could be observed directly during the experiment. If the highest temperature permissible was exceeded, the working surface of the liner exhibited, first, dull-looking streaks from 3 to 10 millimeters wide and extending from one dead center to the other. These streaks appeared distributed at random over the entire circumference of the liner and initiated roughening of the liner. Depending upon wall pressure and temperature, it was possible to observe light streaks for up to half a minute before the actual roughening started. This always began in the dead centers and from these progressively spread from there toward the center of the liner. If the test was immediately stopped as soon as dull streaks appeared, the surface of the ring and the liner disclosed no visible attack. But when perceptible roughening appeared over a thermocouple mounted under the working surface, temperature increases up to 70° C were recorded. The same thing happened if the wall pressure was adjusted too high. With a longer running-in period at lower wall pressure the original

appearance of the rubbing surface could be restored. To ascertain the safe maximum working surface temperatures and wall pressures, oil 6 was subjected to some special tests the results of which are shown in figure 25. Both the ring and the liner being endangered, the tests were confined to 0.4 and 2.7 meters per second average rubbing speeds. As seen, the limits for the maximum wall pressure are much higher as the temperature is lower, and vice versa. The effect of the average rubbing speed is small. The tests, moreover, disclosed the important fact that the wall pressure may not be raised at any working surface temperature above 12 kilograms per square centimeter. (Cf. the plotted test points at 30° and 40° C average working surface temperatures and 13 kg/cm^2 wall pressure.)

PRESSURE CONDITIONS ON THE PISTON RING

The experiments discussed in the foregoing show that the pressure of the piston ring against the rubbing surface of the liner may not exceed 12 kilograms per square centimeter when oil 6 is used, and that this maximum pressure dropped with rising temperature. Next, the experiments by Vogel (reference 11) and by Robertson and Ford (reference 22) showed that the gas pressure in the ring groove at the back of the first (uppermost) piston ring is approximately equal to the pressure in the combustion chamber, and the thought suggests itself whether this gas pressure, which in internal combustion engines may amount to 100 kilograms per square centimeter, should not be looked upon perhaps as wall pressure of the piston ring.

The experiments by Stanton, Mader, and others arrived at no consistent results and were unable to remove the existing confusion. The explanation is facilitated substantially when the pure hydrostatic forces in the lubricating film are strictly differentiated from the forces which in the sense of lubricating technique are due to flow processes in the film or else to a special carrying capacity of adsorbed molecule layers.

Assuming, in connection with the conditions represented in figure 26, that the pressure p_2 below the ring is just as high as the pressure p_1 above it, then purely hydrostatic pressure p_1 prevails in the whole oil film between ring and cylinder wall and the oil film merely has to maintain the equilibrium of the self-expansion of the ring. (If

it were otherwise, it would have been necessary to add in the tests quoted in the present report the atmospheric pressure to the wall pressure of the piston rings.) In the event that p_2 is less than p_1, a linear pressure drop from p_1 to p_2 can be assumed for the data of hydrodynamic friction (that is, of laminar flow in the lubricating film) so that in this instance the hydrostatic pressure amounts on the average to $1/2(p_1 + p_2)$. Hence the lubricating forces in the film must absorb, in addition to the self-expansion pressure p_o of the piston ring, the pressure difference $p_1 - 1/2\,(p_1 + p_2) = 1/2\,(p_1 - p_2)$ - that is, the wall pressure

$$p_a = p_o + \frac{1}{2}\,(p_1 - p_2) \qquad (12)$$

This consideration leads under further simplifications to a very clear representation of the total rubbing force of n piston rings of a piston loaded with the working pressure p_1 (positive pressure). Thus, supposing the friction coefficient μ of all piston rings were the same, the total frictional force becomes

$$R = \mu F(np_o + \frac{1}{2}\,p_1) \qquad (13)$$

It is clear from these two equations that the gas pressure in the combustion chamber effects only in part an increase in the wall pressure of the piston ring. Assuming the pressure below the first sealing ring to be only half as high as above it, as measured by Robertson and Ford (reference 22) on an Otto cycle engine, the uppermost piston ring is loaded only with one-fourth of the gas pressure additionally by the gas.

The question frequently arises in connection with internal-combustion engines whether the piston ring lifts off from the lower groove flank during operation. Eweis has shown that for the two-stroke cycle method no lifting-off may be expected at extremely high rotative speeds in consequence of the mass forces or at missing gas pressure for the last piston ring.

The ring will lift off by reason of the frictional force when the frictional force is greater than the force

with which the ring is pressed by the gas pressure on the lower groove flank. The condition would then be

$$\mu F \left(p_o + \frac{p_1 - p_2}{2} \right) \geqq A p_1 \qquad (14)$$

where

F rubbing area

A side area of piston ring

To illustrate: for $p_1 = 2p_2$ and $\mu = 0.1$, equation (14) becomes

$$p_1 \left(\frac{10A}{F} - \frac{1}{4} \right) \leqq p_o \qquad (15)$$

and if in addition, say, $A = F$ and $p_o = 1.5$ kilograms per square centimeter, the piston ring is lifted off by friction, when

$$p_1 < 0.154 \text{ atmosphere} \qquad (16)$$

It is, therefore, readily apparent that the ring is lifted off from the lower groove flank by the frictional force only at extremely low gas pressures in the combustion chamber.

Translation by J. Vanier,
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for Aeronautics.

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TABLE 2

Oil \ Temp.	44.3°	50.0°	65.3°	77.5°	98.8°	119.6°	149.7°	184.7°	215.0°
1	p_a = 1.5 kg/cm^2								
2		1.5							
3		1.5							
4		1.5							
5	1.5 3.0 4.5 6.0	0.75 1.5 3.0 4.5 6.0	3.0	3.0	3.0	3.0	3.0	3.0	
6			1.5 3.0 4.5 6.0 7.5 9.0	1.5 3.0 4.5 6.0 7.5 9.0	1.5 3.0 4.5 6.0 7.5 9.0	1.5 3.0 4.5 6.0 7.5 9.0	1.5 3.0 4.5 6.0 7.5 9.0	1.5 3.0 4.5 6.0 7.5	1.5 3.0 4.5 6.0
7			1.5 3.0 4.5 6.0	3.0	3.0	3.0	3.0	3.0	3.0
8			3.0	3.0	3.0	3.0	3.0	3.0	3.0
9			1.5 3.0 4.5 6.0				3.0	3.0	

*Test program: The table contains the small pressures maintained at the respective temperature and brand of oil - the average rubbing speed was varied from 0.35 to 3.5 m/sec for each wall pressure. The oil feed in all tests amounted to 150 cm^3/hr, the piston ring employed was ground and sharp-edged.

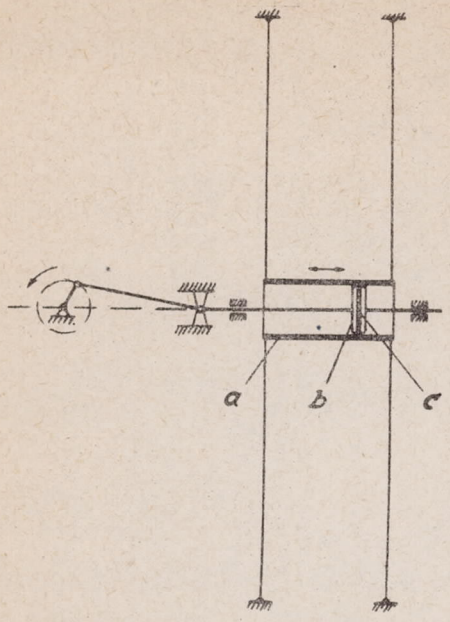


Figure 1.- The original test arrangement. a, liner; b, piston; c, ring.

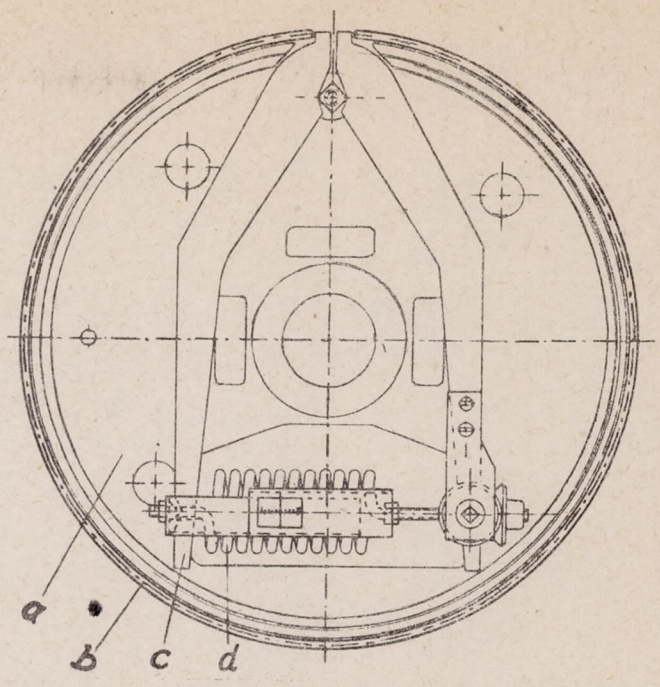


Figure 2.- Spreading device for piston rings. a, piston; b, piston ring; c, spreading lever; d, tension spring with knife-edge suspension.

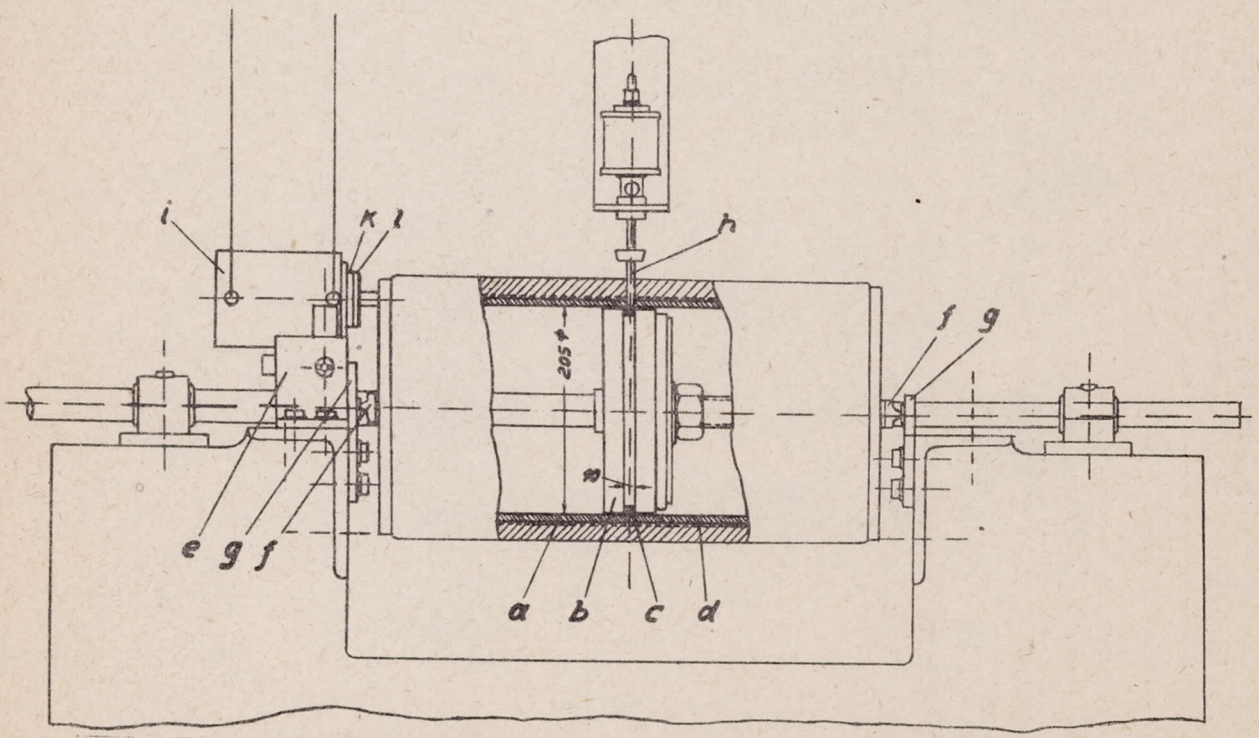


Figure 3.- Test layout. a, liner; b, piston; c, piston ring; d, heating coil; e, arrangement for measuring the deflection of the liner; f, knife edge at the liner; g, spring with socket; h, oil line; i, damping weight; j, ground plates.

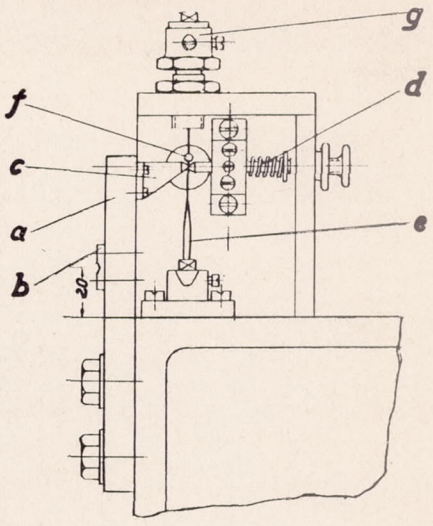


Figure 4.- Mechanical-optical arrangement for measuring the deflection of the liner. a, lengthened spring; b, socket; c, knife edge; d, adjustable knife edge; e, flat spring; f, mirror; g, tension device for spring e.

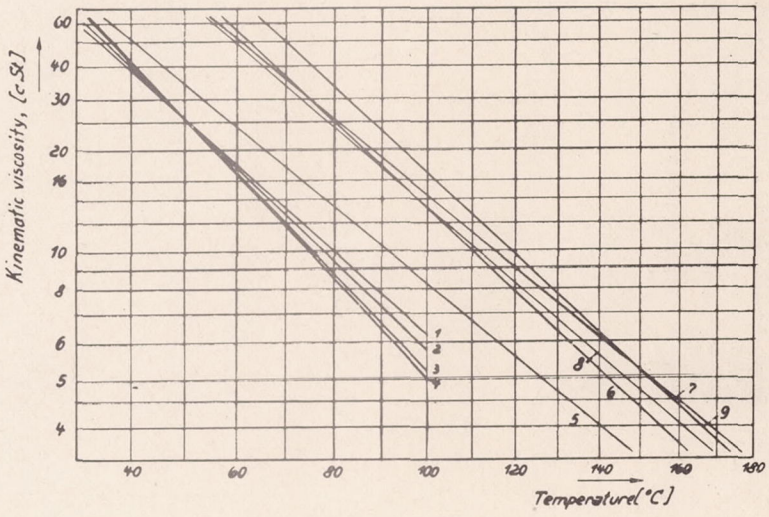


Figure 5.- Temperature relationship of the kinematic viscosity of the test oils.

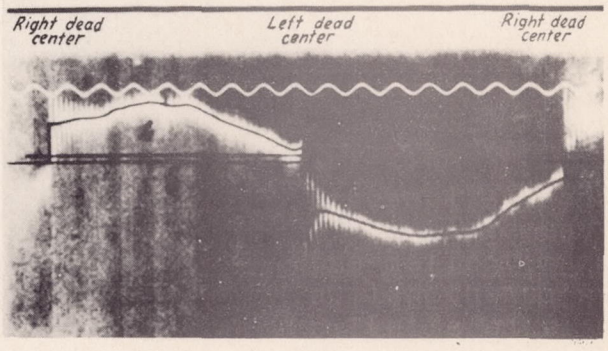


Figure 6.- Force-time record.

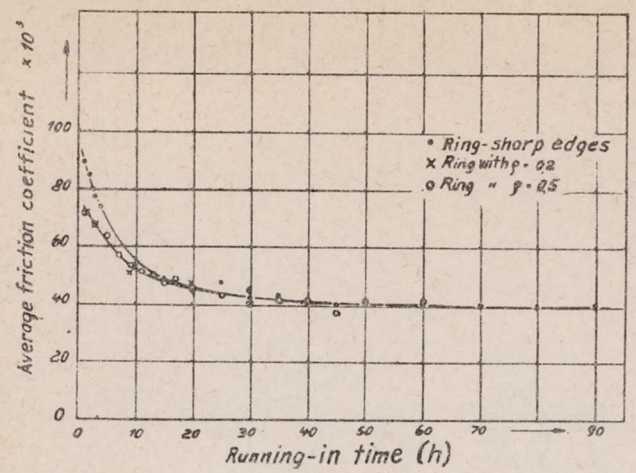


Figure 7.- The average rubbing speed plotted against running-in time. Oil 6- rubbing speed, 2.15 meters per second; wall pressure, 3.0 kilograms per square centimeter; working surface temperature, 120°C.

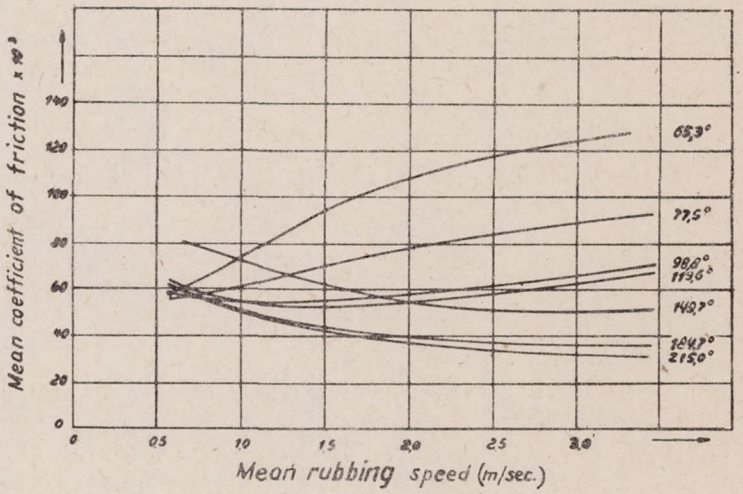


Figure 8.

Figures 8 and 9.- Mean coefficient of friction plotted against mean rubbing speed at different working surface temperatures. Wall pressure, 1.5 and 7.5 kilograms per square centimeter.

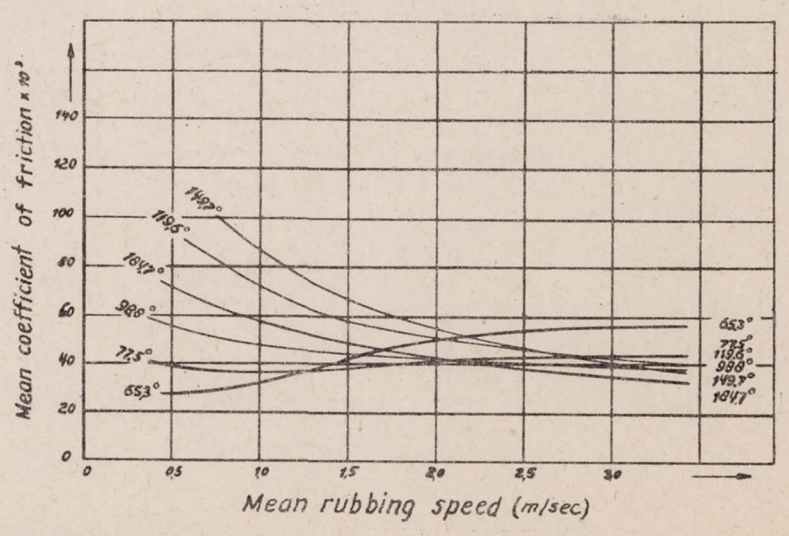


Figure 9.

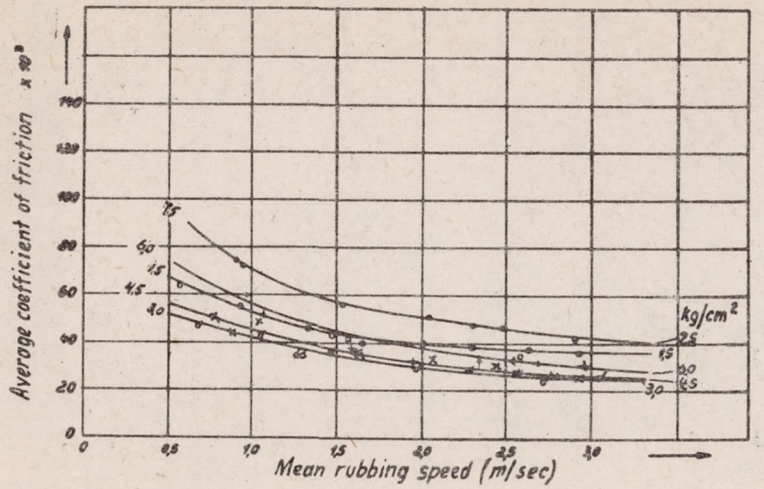


Figure 10.

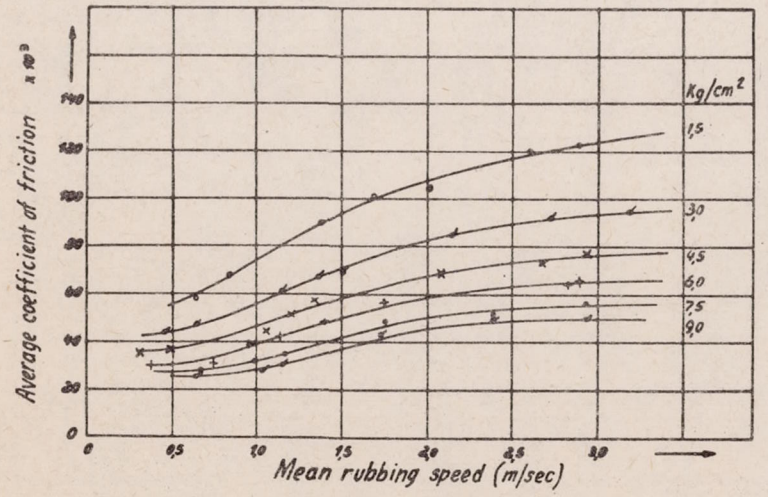


Figure 11.

Figures 10 and 11.- Average coefficient of friction plotted against average rubbing speed at different wall pressures. Working surface temperature, 65.3° and 184.7°C.

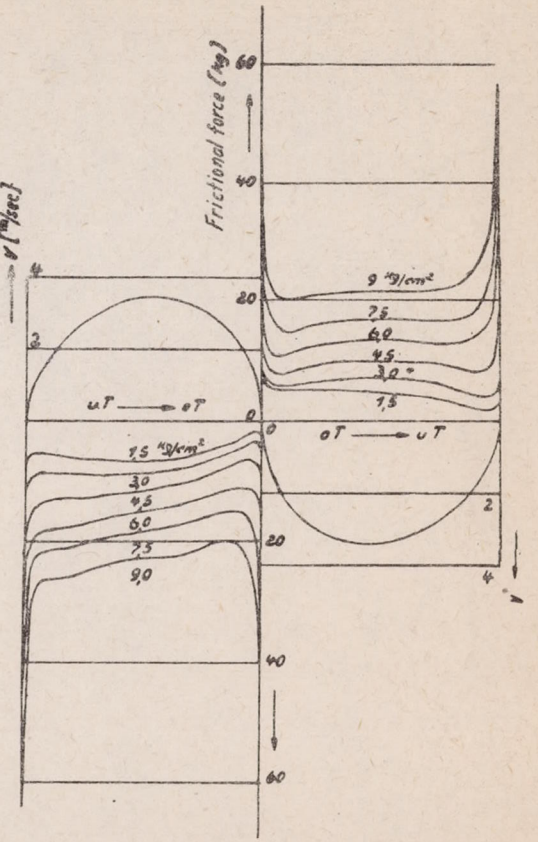


Figure 12.- Oil 6- frictional force plotted against piston setting at different wall pressures. Working surface temperature, 119.6°C; average rubbing speed, 2.15 meters per second; v, momentary rubbing speed.

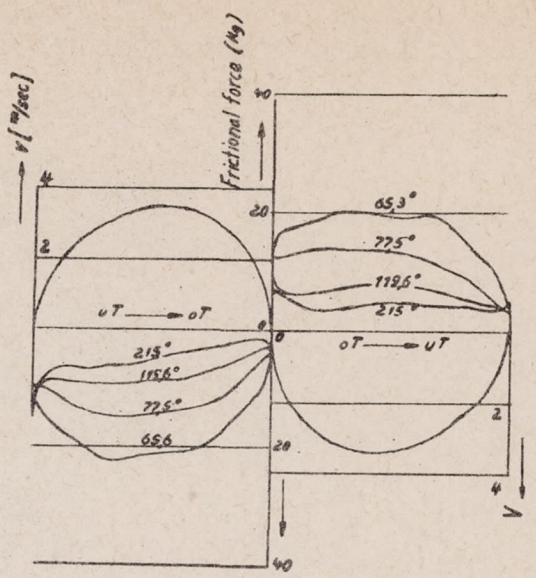


Figure 13.- Oil 6- frictional force plotted against piston settings at different working surface temperatures. Wall pressure, 3.01 kilograms per square centimeter; average rubbing speed, 2.15 meters per second; v, momentary rubbing speed.

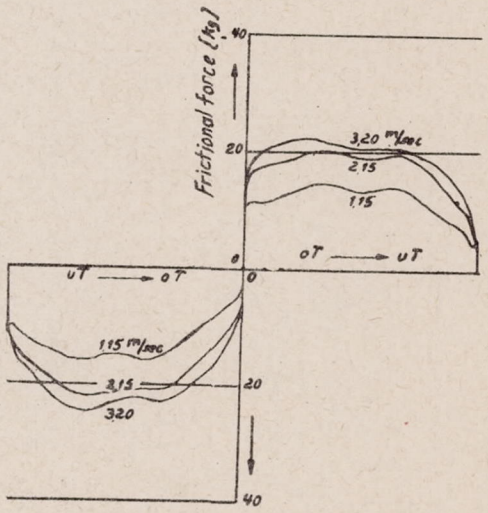


Figure 14.- Oil 6- frictional force plotted against piston speed at different average rubbing speeds. Wall pressure, 3.01 kilograms per square centimeter; working surface temperature, 65.3°C.

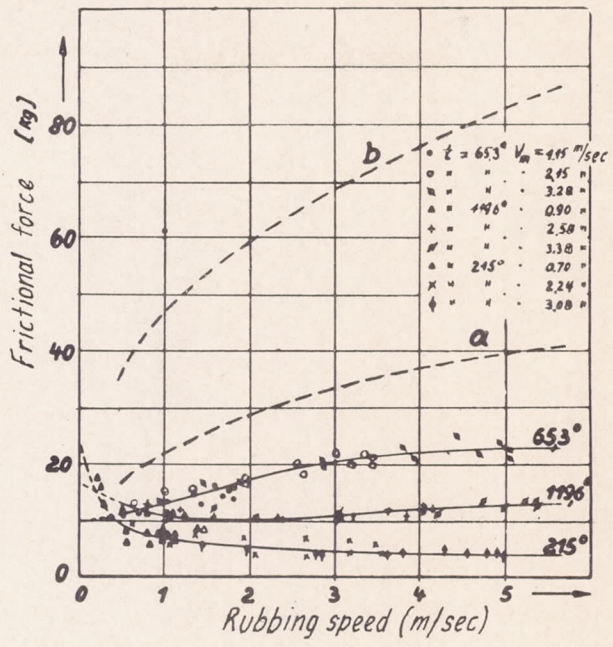


Figure 15.- Oil 6- frictional force plotted against the momentary rubbing speed at different working surface temperatures. Wall pressure 3.0 kilograms per square centimeter, average rubbing speeds, 0.5 to 3.5 meters per second; a and b, the frictional force computed for rounding-off radii 0.1 and 0.01 millimeter at 3.0 kilograms per square centimeter wall pressure and 65.3°C working surface temperature.

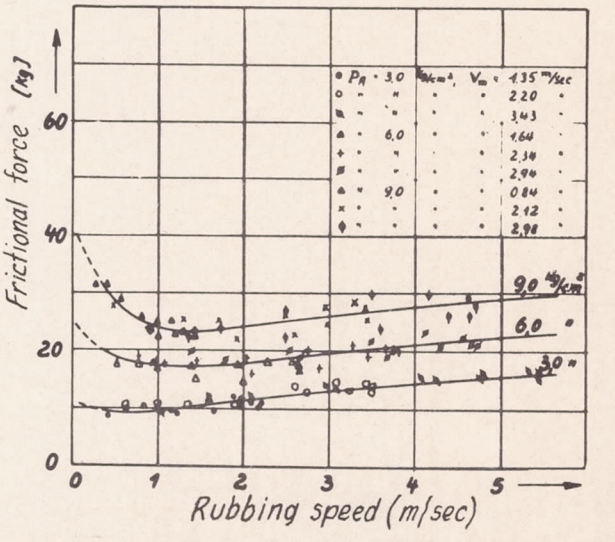


Figure 16.- Oil 6- frictional force plotted against momentary rubbing speed at 3.6 and 9.0 kilograms per square centimeter wall pressures. Average rubbing speeds, 0.5 to 3.5 meters per second; working surface temperature, 77.5°C.

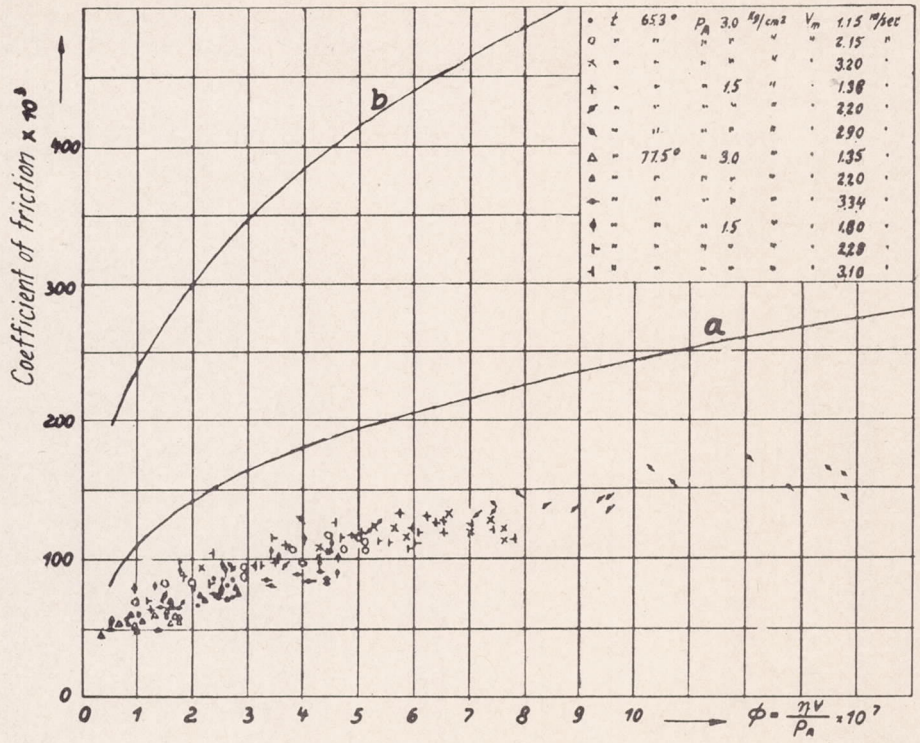


Figure 17.- Coefficient of friction plotted against the characteristic $\varphi = \eta v/p_a$. Curves a and b show the coefficient of friction for 0.1 and 0.01 millimeter curvature radius calculated according to equation (9). The test points reproduce the experimentally obtained coefficient. Test conditions: wall pressure, 1.5 and 3 kilograms per square centimeter; working surface temperatures, 65.3° and 77.5° C; average rubbing speeds, 0.5 to 3.5 meters per second.

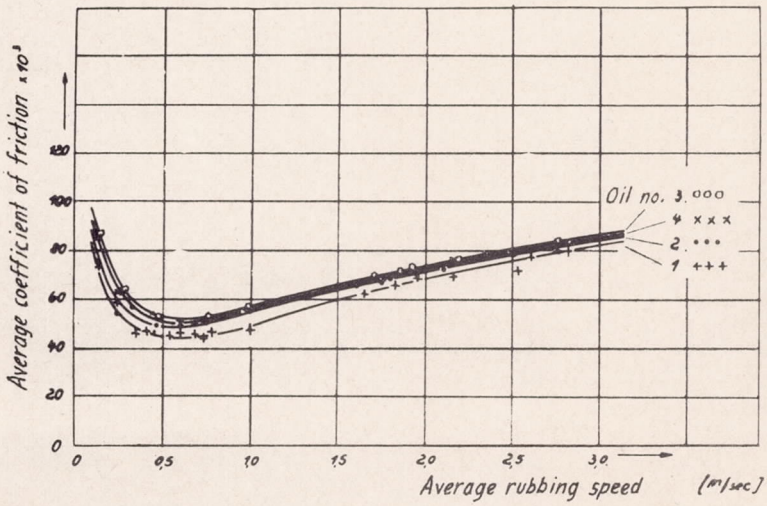


Figure 18.- Oils 1 to 4 having at 50°C test temperature the same absolute viscosity of 0.00230 kilograms per square meter. The average coefficient of friction is plotted against the average rubbing speed. Wall pressure, 1.5 kilograms per square centimeter.

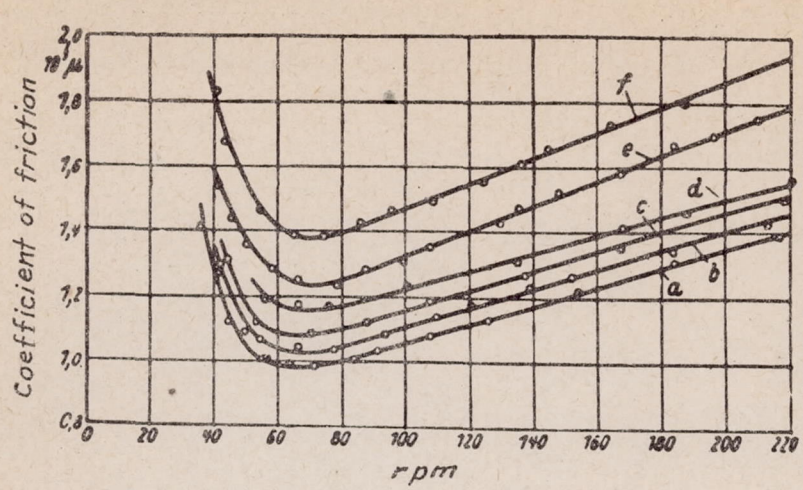


Figure 19.- Coefficient of friction of a journal bearing plotted against rotative speed for different brands of oils (according to Büche).

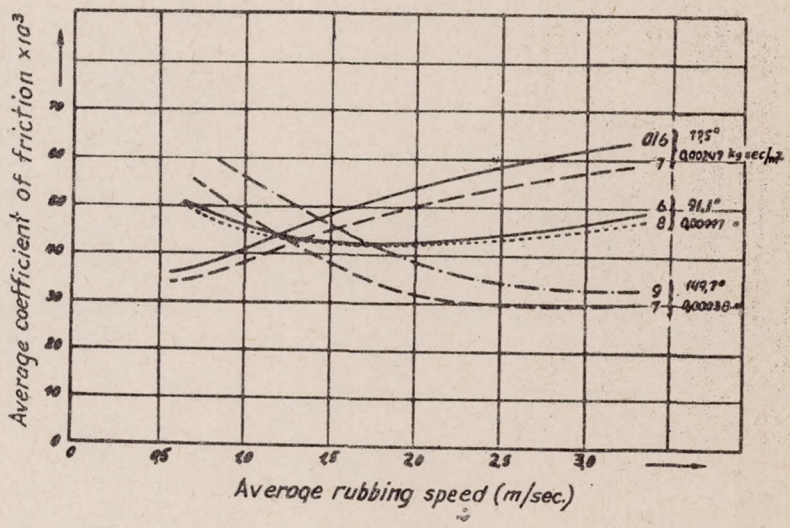


Figure 20.- Oils 6 to 9 having the same viscosity at equal test temperature. Average coefficient of friction plotted against average rubbing speed; wall pressure, 3.0 kilograms per square centimeter.

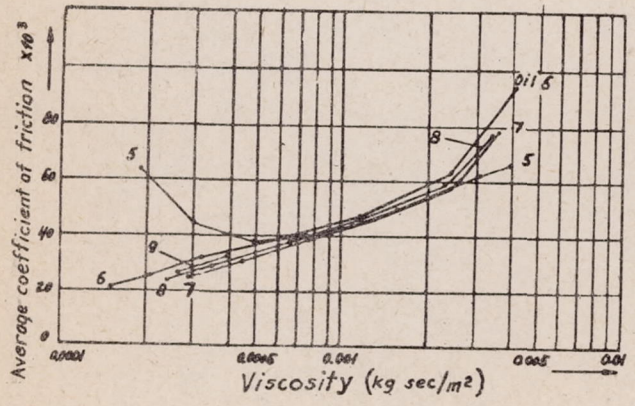


Figure 21.

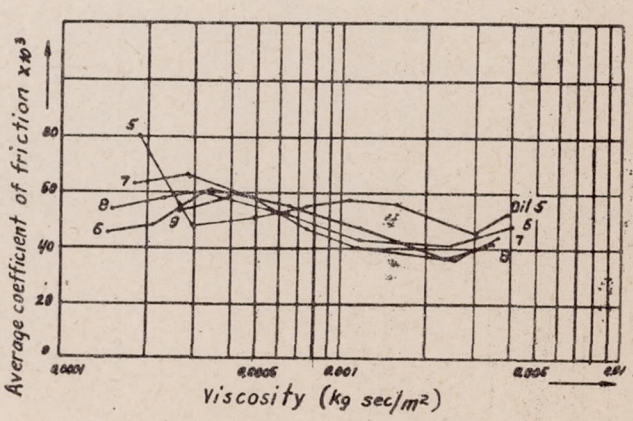


Figure 22.

Figures 21 and 22.- Average coefficient of friction plotted against absolute viscosity. Wall pressure, 3.0 kilograms per square centimeter; average rubbing speed, 3.0 and 0.65 meters per second.

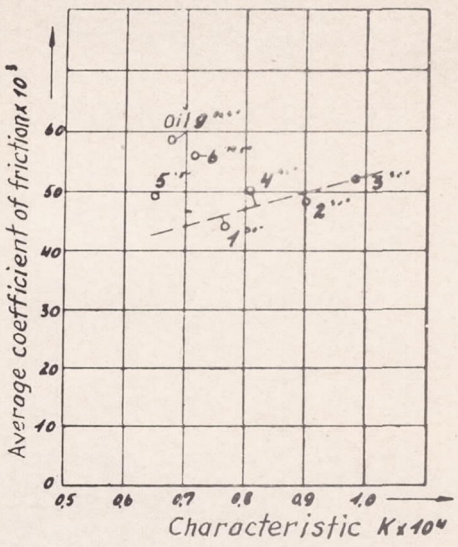


Figure 23.- Average coefficient of friction plotted against the Vogelpohl number. The numerals at the test points indicate the respective type of oil and the pertinent test temperature. Wall pressure, 1.5 kilograms per square centimeter; viscosity, 0.00230 kilogram per square meter; average rubbing speed, 0.6 meter per second; $\eta : \eta_1 = 3 : 1$.

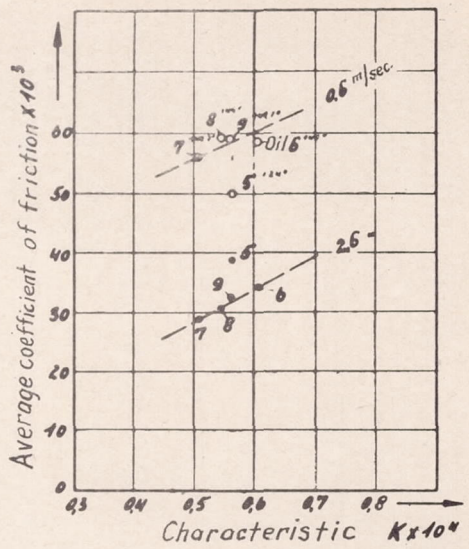


Figure 24.- Average coefficient of friction plotted against Vogelpohl's number. The numerals indicate the respective brands of oil and test temperature. Wall pressure, 3.0 kilograms per square centimeter; viscosity, 0.000380 kilogram per square meter; average rubbing speed, 3.6 and 2.6 meters per second; $\eta : \eta_1 = 1 : 3$.

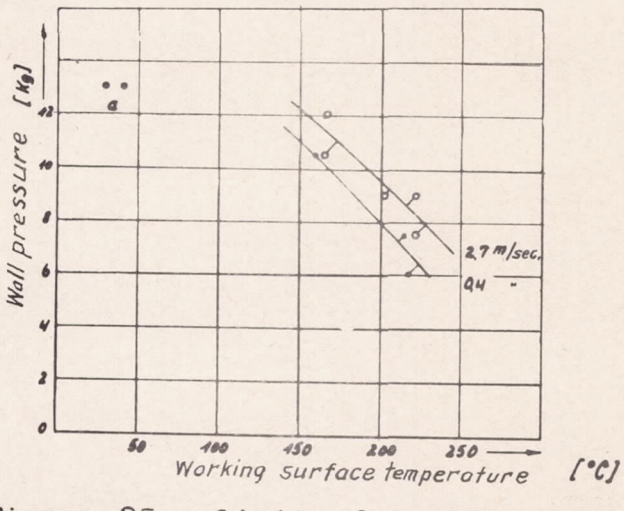


Figure 25.- Limits for wall pressure and working surface temperature on oil 6. Average rubbing speed, 0.4 and 2.7 meters per second; at test points a the roughening appeared even during starting.

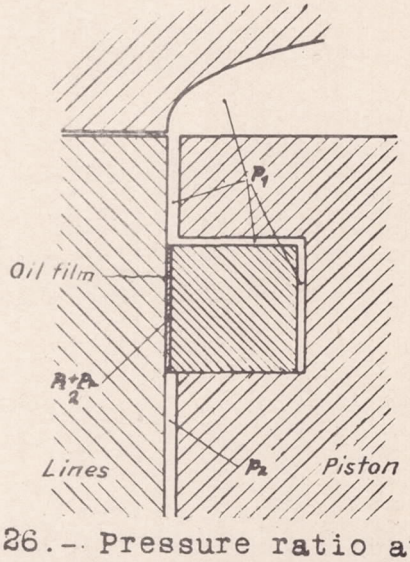


Figure 26.- Pressure ratio at the piston ring. p_1 = gas pressure in combustion chamber above and back of the ring; p_2 = gas pressure below the ring.